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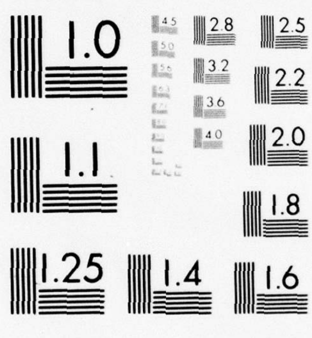

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TEST OF DYNAMIC FORCE THRUST CONVECTOR HEAT EXCHANGER FOR 20 KW COMPACT ARC XENON LAMPS

Final Technical Report

Author: James R. Palmer

With Editing and Supplement by The Technical Staff,
Aerospace Controls Corporation

24 October 1968

U. S. Army Mobility Equipment
R. & D. Center Fort Belvoir, Va.
DAAK02-68-C-0470

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I. Introduction

A. Purpose

The purpose of the program covered by U. S. Army Mobility Equipment R & D Center, Contract Number DAAK02-68-C-0470 was to establish by laboratory test the feasibility of air cooling anodes for 20 KW xenon short arc lamps.

B. History

Xenon short arc lamps in power ratings to about 6.5 KW use a flow of air longitudinally along the lamp envelope to carry off heat generated during lamp operation. Heat generated at the arc gap is conducted by the metallic electrodes along their length: the electrodes have ends which are exposed to the air flow, and the rapidly-circulating xenon gas within the envelope also picks up heat from the electrodes. The gas is cooled by contact with the relatively cool surface of the lamp quartz envelope. Additional heat is dissipated by radiation to surrounding materials.

It was found that this cooling mechanism is not satisfactory at power levels of about 8 KW: the rate of heat transfer is inadequate to prevent melting of electrode materials accompanied by darkening of the lamp caused by deposition of vaporized metal on the inside of the lamp envelope, and unacceptably short lamp life. The effect is especially noted at the anode: the direction of the direct current from cathode to anode causes a bombardment of the anode surface: the heavy concentration of electrons induces a high heat load at the anode surface. Additionally it is desired to maintain a compact arc for reasons of optical efficiency in the collection and projection of the light energy: in turn, this causes the high heat load to be applied

over a small area. Thus the problem is one of relieving a high heat density applied to a small surface.

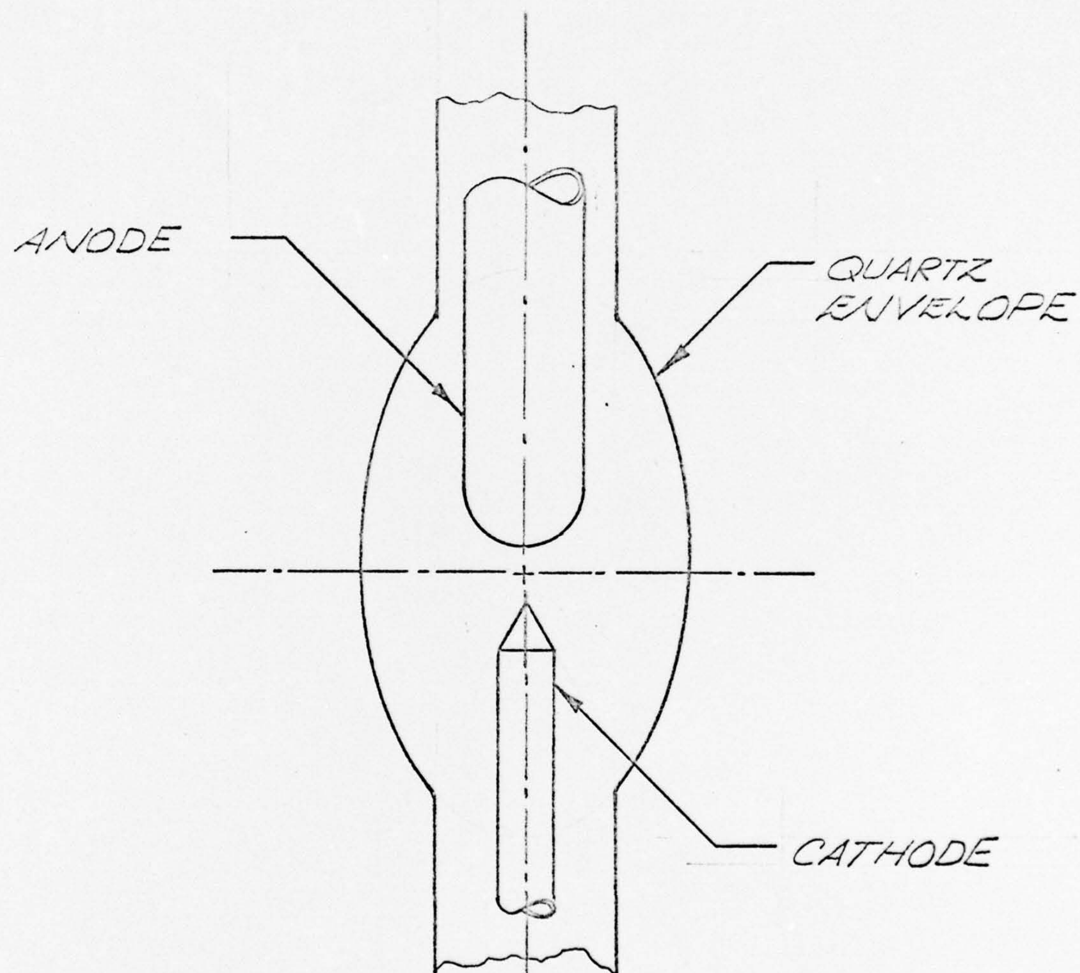
Figure 1 shows a schematic of a typical short arc xenon lamp. Attention is called to the configurations of the anode and of the cathode: the cathode usually has a pointed tip to conduct the current in the smallest possible cross-section area, and thus create a bright "fireball" across the gap. The anode presents a hemispherical face to the current so as to spread the heat loading over a relatively large area. Although the flame develops a "tail", or spreads out over the anode, the largest heat concentration is in a small area directly ahead of the cathode tip.

For the above reasons the development of the 20 KW short arc xenon lamp necessitated development of an alternate method of cooling, especially cooling the anode. The designs that proved best use hollow anodes or anodes with passages through which water, or some other liquid coolant, is forced at high pressure and at high velocity.

With water it was noted that nucleate boiling occurred at the anode tip, and even with distilled water impurities present in the water would deposit at the tip. These deposits created an insulation which interfered with heat transfer, thus overheating and early failure of the anode and of the lamp.

A number of efforts were made to further isolate the problem; to locate a coolant which would comply with the high and low ambient temperature conditions encountered when these lamps are used in Army applications, or to devise some approach that would solve the problem in a practical manner.

Aerospace Controls Corporation made an analytical study of the use of air cooling of these hollow-anode lamps. The study showed that a



20KW LAMP

FIGURE 1

direct air blast through the anode passages was not sufficient for cooling: the volume of air required was such that supersonic flow was needed, and at speeds above that of sound the air flow would add heat to the surface rather than removing it. However, a continuation of the study showed that if a coiled tube were inserted into a hollow anode; and if the remainder of the cavity were filled with a liquid material which would absorb heat from the inside of the anode body and transfer that heat to the tube; then it would be possible to force air through the tube in sufficient volume and at subsonic velocity to conduct the heat away from the anode body. The mechanism of heat transfer would be force convection: the liquid medium would naturally circulate within the cavity and during circulation would absorb heat from the anode and transfer it to the air-cooled coiled tube.

As a preliminary test, a length of copper tubing was coiled into an approximation of the tube that would be used in a hollow anode for a 20 KW lamp. One end of this tube was connected to an air compressor line, the other end exposed to free air. The tube was immersed in a container of boiling oil, to which heat was continuously applied by a gas burner, and the air was turned on. It was noted that when the air was turned on the boiling action ceased; when the air was turned off the boiling started again after a short time: the interval before re-boiling varying with the length of time the air was on.

The set-up was not instrumented to measure the extent of heat transfer; however, a temperature rise of about 75 degrees was noted in the air, and the results as evidenced by the boiling action established that heat transfer was taking place.

The analytical study was presented to the Army together with a recommendation that a laboratory experiment, to test the device, be undertaken.

The experiment would verify or refute the analysis that force convection would conduct the heat away under conditions approximating the high density heat loading characteristic of 20 KW short arc xenon lamp anodes.

II. Test Program

A. Analysis

In the course of developing compact plasma arc lamps, the concept of cooling anodes by forced convection was avoided for a very basic reason, viz., the anode configuration, and the thermal loading, which is concentrated on a very small surface area, does not permit sufficient area of contact relative to the amount of cooling fluid to obtain high Reynold's numbers for forced convection. Recent advances have been made, however, by several lamp manufacturers in utilizing a venturi effect in cooling the anode tip. Antecedent to these recent advances, the primary heat transfer mechanism for removing the high thermal loads was nucleate boiling.

The nucleate boiling heat transfer mechanism is best described as utilizing the heat of vaporization at a hot spot to cool that spot and to force boiling independent of fluid velocity. The agitation caused by the bubbles is more effective than turbulence in forced convection without boiling. Succinctly stated, nucleate boiling is dependent upon high heat flux in order to produce the bubble regime, in order to provide proper evaporative cooling. The essence of the phenomena, relative to the problems encountered are:

Excessive velocity of the fluid decreases the surface tension and raises the boiling point of the fluid.

Hydrodynamic crisis occurs when the velocity of the liquid, relative to the velocity of the vapor coming from the heating surface, is so great that a further increase would either cause the vapor columns to drag the liquid away from the heating surface, or the liquid streams to drag the vapor back towards the heating surface. Consequently, there is a limiting relationship between the heat flux and flow condition which is quite critical.

Pressure changes the density of the saturated vapor and of the saturated liquid and inhibits the process of vapor removal from the heating surface by causing the bubble removal to be intermittent instead of continuous, thereby acting as an insulator.

Pressure also affects the peak heat flux because it changes the vapor density and therefore the boiling point. It follows, then, that the boiling point affects the heat of vaporization.

Other problems, familiar to those who have had experience in lamp operation and development include the selection of shape of the heating surface, and the coolant material to be used. In the latter area, coolants which will withstand extreme low temperatures may chemically break down at high thermal fluxes if not properly inhibited. Other materials have too high a boiling point for evaporative cooling. Water displays the best characteristics in terms of latent heat of evaporation, density, surface tension and specific heat, but freezes at too high a temperature and, because it is such a good solvent, can carry ionized impurities which can greatly affect the metallurgical properties of the anode. Impurities deposited as a result of nucleate boiling of water further insulate the heating surface and degrade cooling capabilities. Further, metallic ions can be forced into chemical combinations with

the anode metallic structures causing a degradation in the anode structure and hence can cause thermal stresses, fatigue fracture, and subsequent grain growth.

For a forced convection heat transfer mechanism to be successful, it is necessary to spread the thermal load and more uniformly distribute the thermal flux in the thermal vehicle.

In order to more uniformly distribute the thermal load, we have endeavored to develop a two stage compact heat exchanger. This system is denominated as a Dynamic Force Thrust Convector.

Basically, the first stage allows the heat load from the lamp anode to free conduct to the fluid in the isolated thermal-well. The fluid is caused to force convect and more uniformly distribute the load through the body of the fluid. Consequently, the cooling heat exchanger chamber will then pick up the thermal load from the fluid in the first stage. The thermal-well fluid will achieve a point of equilibrium with the first stage fluid and provide the necessary Δ_T across the anode to provide conduction through the anode wall at the thermal equilibrium point.

At the outset, the first stage medium was to remain encapsulated, with a provision for expansion. However, because of the propensity to emulsify on the part of the Methyl Phenol Siloxane, it became necessary to provide a kinetic property to the first stage. The second stage was to utilize a fluid, either compressible or incompressible, as the medium to remove heat from the first stage. The two second stage materials elected were compressed air and water.

The underlying concept for air cooling is based in large measure on Diabatic Flow, to wit: Ref. 1)

1. In subsonic flow, velocity and Mach number increase and the pressure and density decrease in the direction of flow.

2. The temperature increases if the Mach number is less than $\frac{1}{\sqrt{K}}$ (K = 1.40 for air).

3. Subsonic flow approaches sonic condition when heat is added; but, it cannot go beyond subsonic, i.e., it cannot become supersonic.

Measurements of anode heat loading of 20 KW short-arc xenon lamps showed that the anode is receiving approximately 8KW.

$$8.0 \times 10^3 \text{ Watts} \times 3.413 \ \frac{\text{BTU/HR}}{\text{WATT}} = 2.73 \times 10^4 \ \frac{\text{BTU}}{\text{HR}}$$

<u>A Priori</u> statements

1. The inlet gas temperature = 60°F., i.e., 520°R.

2. The inlet pressure = 124.7 PSIA, i.e.

Therefore, the density $\frac{\#}{\text{Ft.}^3}$ of air at the <u>a priori</u> values is: (Ref. 3)

$$\frac{1.44 \times 10^2 \ (1.247 \times 10^2)}{\frac{1.544 \times 10^3}{2.9 \times 10} \ (5.20 \times 10^2)} = 6.24 \times 10^{-1} \ \frac{\#}{\text{Ft.}^3} \qquad 1.$$

The required $\frac{\text{Ft.}^3}{\text{Hr.}}$ required to unload 2.73×10^4 $\frac{\text{BTU}}{\text{Hr.}}$ follows as:

$$\frac{\text{Ft.}^3}{\text{Hr.}} = \frac{2.73 \times 10^4 \ \frac{\text{BTU}}{\text{Hr.}}}{6.24 \times 10^{-1} \ \frac{\#}{\text{Ft.}^3} \ (2.40 \times 10^{-1} \ \frac{\text{BTU}}{m\text{-F}^o}) \ (2.0 \times 10^{2\,{}^oF.} \ \triangle T)} \qquad 2.$$

$$= 9.11 \times 10^2 \ \frac{\text{Ft.}^3}{\text{Hr.}}$$

The air will flow through a .375 in. O.D. x .311 in. I.D. coiled tube.

Therefore, the initial velocity (denoted V_1) will be:

$$V_1 = \frac{\dfrac{9.11 \times 10^2 \ \frac{Ft^3}{Hr.}}{3.60 \times 10^3 \ \frac{Sec.}{Hr.}}}{\dfrac{\pi \left(\frac{3.11 \times 10^{-1} \text{ in.}}{2}\right)^2}{1.44 \times 10^2 \ \frac{in.^2}{Ft.^2}}} \qquad 3.$$

$$= 4.78 \times 10^2 \ \frac{Ft.}{Sec.}$$

It is necessary to evaluate the amount of forced convection which can be translated from the fluid in the anode, in which the coiled tube is immersed. Taking the average film temperature, (Ref. 1 and 5) i.e.,

$$T_f = 0.5 \ (T_{surf} - T_{bulk}) = 1.30 \times 10^2 \ {}^{o}F.$$

$$\therefore \ \bar{h}_c = 2.3 \times 10^{-2} \left[Re_{D_f}\right]^{0.8} \left[Pr_f\right]^{0.33} \left[\frac{k_f}{D}\right] \qquad 4.$$

where:

$$Re_{D_f} = \frac{VD\rho}{\mu}$$

$$V = 4.78 \times 10^2 \ \frac{Ft.}{Sec.}$$

$$D = 2.59 \times 10^{-2} \ Ft.$$

$$\rho = 6.24 \times 10^{-1} \ \frac{\#}{Ft.^3}$$

$$\mu = 1.30 \times 10^{-5} \ \frac{\#m}{Ft. - Sec.}$$

$$Re_{D_f} = \frac{4.78 \times 10^2 \ \frac{Ft.}{Sec.} \ (2.59 \times 10^{-2} Ft.)\ (6.24 \times 10^{-1} Ft.)}{1.30 \times 10^{-5} \ \frac{\#m}{Ft. - Sec.}}$$

$$Re_{D_f} = 5.95 \times 10^5$$

$$Pr_f = \frac{cp \ \mu}{K} \ (3.60 \times 10^3 \ \frac{Sec.}{Hr.})$$

$$Pr_f = 7.2 \times 10^{-1}$$

$$c_p = 2.40 \times 10^{-1} \ \frac{BTU}{\#m - {}^oF.}$$

$$\mu = 1.30 \times 10^{-5} \ \frac{\#m}{Ft. - Sec.}$$

$$K = 1.60 \times 10^{-2} \ \frac{BTU}{Hr.-Ft.-{}^oF.}$$

Hence:

$$\bar{h}_c = 2.3 \times 10^{-2} \left[5.95 \times 10^5\right]^{0.8} \left[7.2 \times 10^{-1}\right]^{0.33} \left[\frac{1.60 \times 10^{-2} \ \frac{BTU}{Hr-Ft-{}^oF.}}{2.59 \times 10^{-2} \ Ft.}\right]$$

$$\bar{h}_c = 5.35 \times 10^2 \ \frac{BTU}{Hr.-Ft.^2 - {}^oF.}$$

Correction Factor for Temperature

$$St = \frac{2.1 \times 10^{-2} \left[R_e\right]^{-0.2} \left[\frac{T_b}{T_s}\right]^{0.575}}{Pr^{0.66}}$$

$$St = 8.5 \times 10^{-4}$$ 5.

Therefore:

$$\bar{h}_c = 2.40 \times 10^{-1} \ \frac{BTU}{\#m - {}^oF.} \ 5.04 \times 10^{-1} \ \frac{\#}{Ft.^3}) \ (8.44 \times 10^2 \ \frac{Ft.}{Sec.})(3.60 \times 10^3 \ \frac{SEC}{Hr.}) \ \times 8.5 \times 10\text{-}4$$

$$\bar{h}_c = 3.12 \times 10^2 \ \frac{BTU}{Hr.-Ft.^2 - {}^oF.}$$

The area of surface contact with the immersed fluid is found to be:

Pitch:

$$L = .375 + .010$$

$$= .3850$$

Overall length:

$$H = N\ L = 3d$$

$$\therefore N = \frac{8.0 \text{ in.} - 1.125 \text{ in.}}{.3850 \text{ in.}}$$

$$N = 17.85$$

$$l = \left[\frac{1.25 \text{ in.}}{\text{coil}} (1.785 \times 10 \text{ coils})\right]\pi$$

$$= 70.25 \text{ in.}$$

$$A_T = \frac{\left[\pi\ (3.750 \times 10^{-1} \text{ in.})\right]\left[70.25 \text{ in.}\right]}{1.44 \times 10^{2}\ \frac{\text{in.}^2}{\text{Ft.}^2}}$$

$$= 8.2 \times 10^{-1} \text{ Ft.}^2$$

Consequently, knowing the heat transfer coefficient and the area of contact, the amount of heat that can be "off-loaded" by forced convection follows:

$$\dot{q}_c = 3.12 \times 10^2\ \frac{\text{BTU}}{\text{Hr.-Ft.}^2\text{-}^{\circ}\text{F.}}\ (8.2 \times 10^{-1} \text{ Ft.}^2)\ (2.0 \times 10^2\ ^{\circ}\text{F.}_{\Delta T}) \quad 6.$$

$$= 51.2 \times 10^3\ \frac{\text{BTU}}{\text{Hr.}}$$

Since the air can absorb $2.73 \times 10^4\ \frac{\text{BTU}}{\text{Hr.}}$ only, then the $\Delta T_{^{\circ}\text{F.}}$ will be 107°F. indicating that the wall surface tube will be = 167°F. This is, of course, neglecting the other two mechanisms of $\dot{q}_k$ (conduction) and $\dot{q}_R$ (radiation).

The temperature of the fluid in the isolated thermal well will be $\cong 167^{\circ}F$. The translation of heat from the Xe side to the fluid is by forced convection of the fluid in the thermal well and straight conduction. As a result, the heat will transfer from the Xe side to the fluid in the thermal well is according to the following method:

$$q_k = \frac{K\ (A)}{L}\ (\Delta T^{\circ}F.) \qquad 7.$$

where:

$$K_{Cu} = 2.24 \times 10^{2}\ \frac{\text{BTU}}{\text{Hr.-Ft.}^{2}\text{-}^{\circ}\text{F.}}$$

$$A = \frac{\left[\frac{4\ (\pi)\ (6.25 \times 10^{-1}\ \text{in.})^{2}}{2}\right]}{1.44 \times 10^{2}\ \frac{\text{in.}^{2}}{\text{Ft.}^{2}}} \qquad L = 2.5 \times 10^{-3}\ \text{Ft.}$$

$$2.73 \times 10^{4}\ \frac{\text{BTU}}{\text{Hr.}} = \frac{2.24 \times 10^{2}\ (1.63 \times 10^{-2})}{2.5 \times 10^{-3}}\ (x - 167^{\circ}F.)$$

$$x^{\circ}F. = \frac{2.73 \times 10^{4}\ \frac{\text{BTU}}{\text{Hr.}}\ (2.5 \times 10^{-3}\ \text{Ft.})}{3.658\ \frac{\text{BTU}}{\text{Hr.-}^{\circ}\text{F.-Ft.}}} + 167^{\circ}F.$$

$$= 18.69 + 167^{\circ}F.$$

Xe side of Anode Temperature $\cong 185^{\circ}F$. with a wall thickness of .030 in. While the above analysis is predicated upon the use of Methyl Phenol Siloxane, pour point - $100^{\circ}F$., it does not preclude the use of other materials which are equal or superior in qualities and performance.

The flow in the tube can reach a maximum Mach number (M^{Y}). At the limiting point, the pressure gradient is infinite. Consequently, the differential expression:

$$\frac{dp}{dx} = \frac{\frac{Pf}{2D}}{1 - \frac{P}{\rho V^{2}}} = \frac{\frac{f}{D}\frac{\rho V^{2}}{2}}{k M^{2} - 1} \qquad 8.$$

which results in $M^{Y} = \frac{1}{\sqrt{k}}$

therefore, the velocity increases along the tube until it reaches $\frac{1}{\sqrt{k}}$ which is .845 for air.

The initial M_1 number is:

$$M_1 = \frac{7.0 \times 10^2 \ \frac{Ft.}{Sec.}}{\sqrt{kg\ RT}} = \frac{7.0 \times 10^2 \ \frac{Ft.}{Sec.}}{1.118 \times 10^3 \ \frac{Ft.}{Sec.}} \qquad 9.$$

$$M_1 = .627$$

The pressure drop expected for a run of tubing 15' - 0" long: (Ref. 6 and 7)

$$1 - \left(\frac{P_2}{P_1}\right)^2 = k\ M_1 \left(F \frac{L}{D} + 2 \ln \frac{P_1}{P_2}\right) \qquad 10.$$

$$F = \frac{1}{\sqrt{\lambda}} = 2.0 \log \left(Re_D \sqrt{\lambda}\right) - 0.8$$

$$F = 1.31 \times 10^{-2}$$

$$P_2 = 93.4$$

$$V_2 - \frac{V_1 \ P_1}{\rho_2}$$

$$V = 9.87 \times 10^2 \ \frac{Ft.}{Sec.}$$

$$M_2 = \frac{9.87 \times 10^2 \ \frac{Ft.}{Sec.}}{1.118 \times 10^2 \ \frac{Ft.}{Sec.}}$$

$= .884$ (uncorrected for coil resistance.)

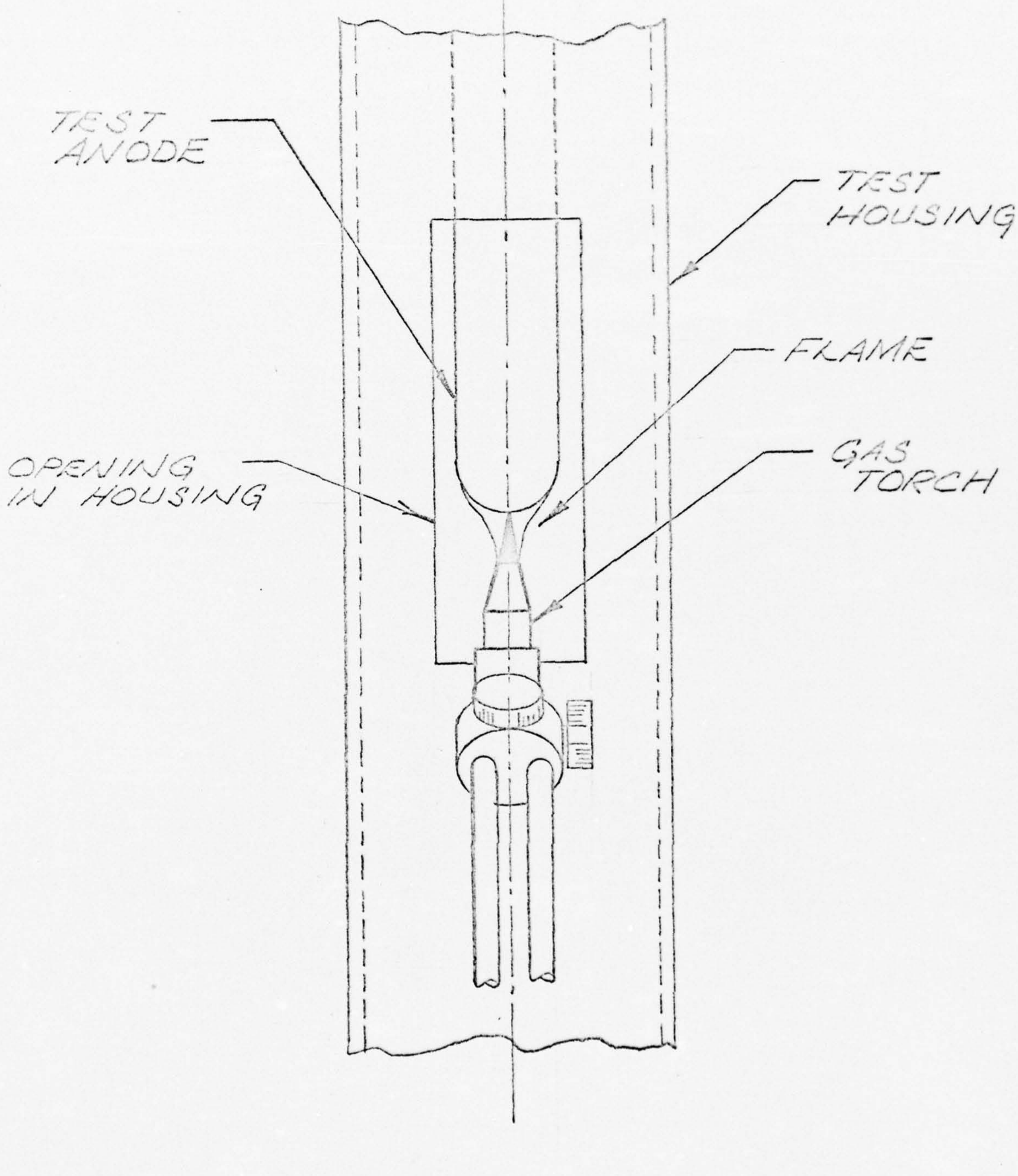
B. Test Program

B.1 Set-up: Test No. 1

A schematic of the test apparatus is shown in Figure 2, and photographs in Figure 3. The three-inch diameter steel tube forms a shield about the anode which simulates the lamp envelope, and simulates the gas heat dynamics found in the 20 KW lamp. Heat loading on the tip of the anode was simulated using an oxy-acetylene torch with a Victor #5 cutting tip. The rate of heat application was controlled by control of the gas pressure and flow. The area over which the concentrated heat from the oxy-acetylene flame was to be applied was controlled by placement of the flame tip, i.e., moving the torch nearer to or further away from the test part.

Experience with characteristics of burned lamp anodes shows that, in general, the area of damage is approximately in line with the arc centerline, with melting of the anode tip metal starting at a centerpoint and spreading out to a diameter of about 2 millimeters. Therefore, in order to simulate the highly concentrated heat loading the oxy-acetylene flame tip was set directly at the anode surface as sketched in Figure 2.

In order to proceed into the program in an orderly manner, and to minimize the possibility of damage to the test parts, it was decided to start out on a reduced-severity basis. Two items were introduced: 1) A propeller driver by a small electric motor was inserted through the center of the anode coil. The intent of the impeller was to provide a positive means of circulating the liquid medium in the



TEST SET-UP

FIGURE 2

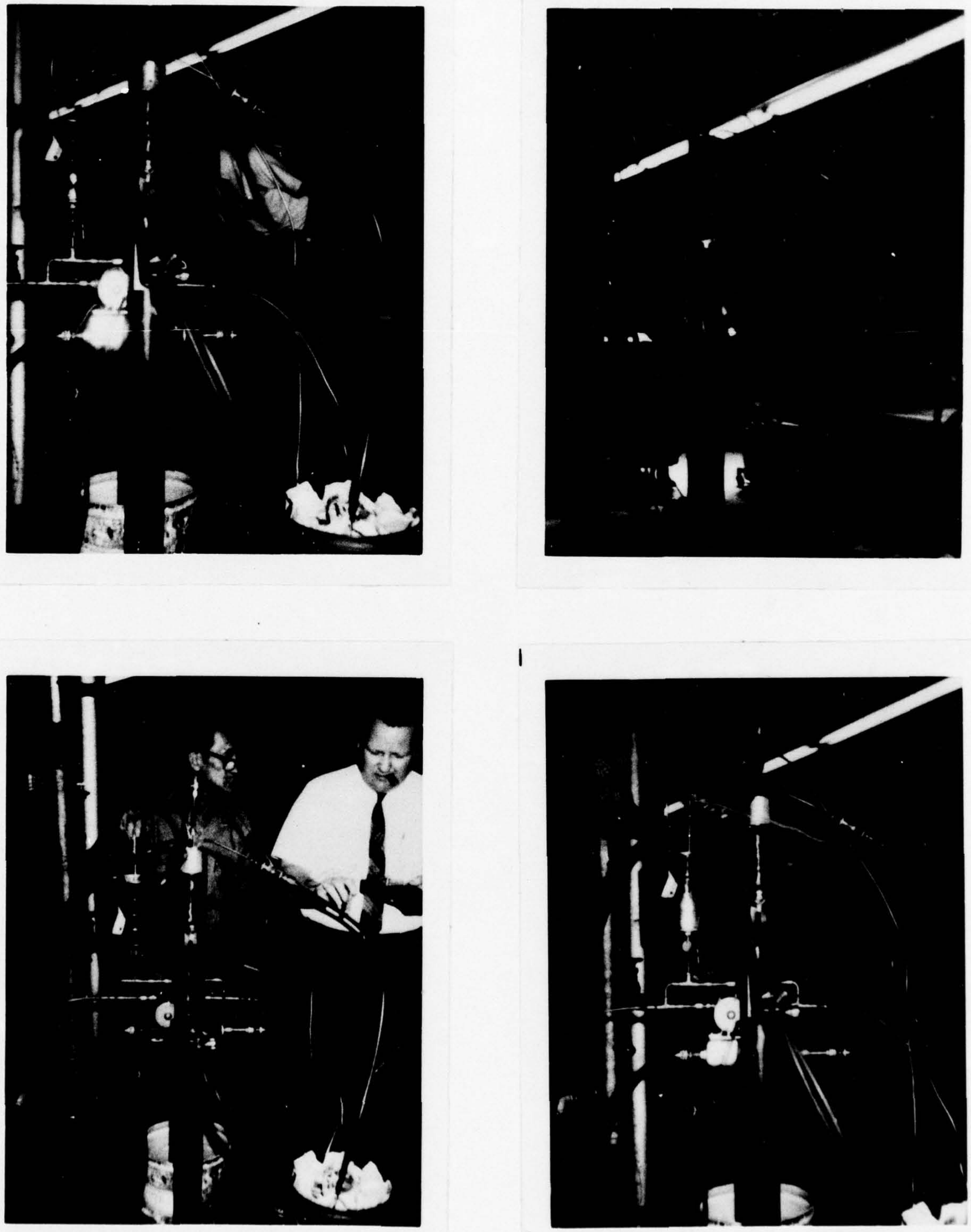


FIGURE 3

anode: the impeller was driven so as to draw the fluid in an upward direction along the impeller shaft. The arrangement is shown in Figure 4.

2) Because of improved heat transfer characteristics, water, rather than air, would be circulated through the cooling coil.

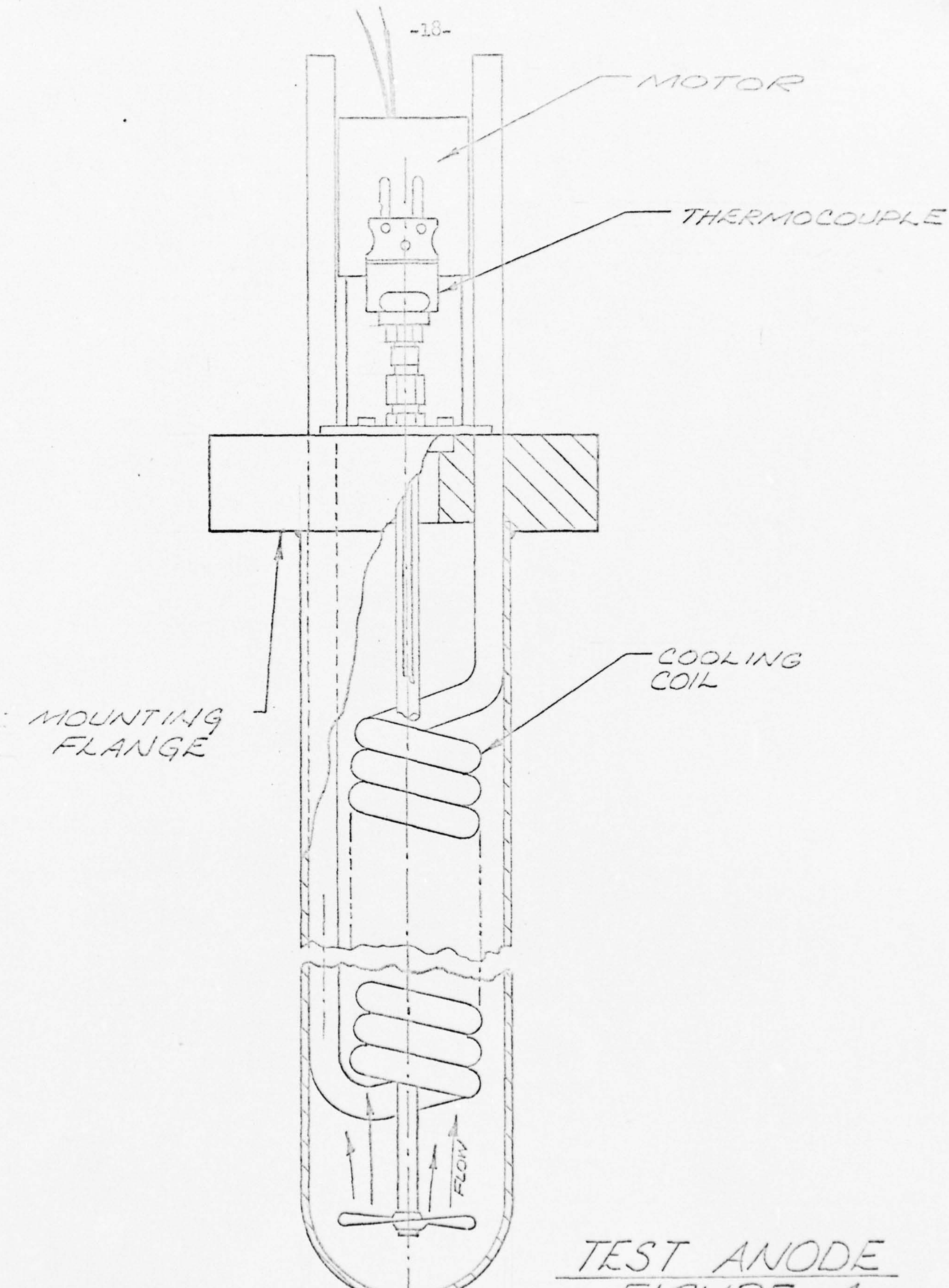
It was planned that as testing proceeded, and if results were in accordance with the analysis, the impeller would be removed and/or air would be substituted for the water.

Thermocouples were placed within the anode cavity, and at the water inlet and outlet lines immediately adjacent to the coil, a flowmeter was placed in the water inlet line. Acetylene gas pressure was adjusted and monitored to permit calculation of gas flow through the torch tip orifice and heat applied to the anode. A surface thermocouple probe with read-out was used to test surface temperatures at various areas of the anode body.

The test anode duplicated the size and shape of the anode used in the Hanovia 20 KW lamp. The anode was manufactured of copper: there was no tungsten used at the tip; and the wall thickness at the anode tip was .070 inch.

The cooling coil used 3/8" O.D. by .156 wall copper tubing coiled to an outside diameter of one inch, and with .025 inch spacing between the coils. The general arrangement of the test anode is shown on drawing Figure 4 following.

The anode cavity was filled with Methyl Phenol Siloxane.



TEST ANODE
FIGURE 4

B.2. Results: Test No. 1

The cooling water flow was turned on, the propeller was energized, all thermocouples were connected to their respective read-out instruments, and a flame, set somewhat below the pressure level calculated to apply rated heat (8 KW, or 27,300 BTU/hr.), was applied to the test anode. It was noted that the temperature of the Methyl Phenol Siloxane rose rapidly, but that there was only a small rise in temperature of the cooling water. Cooling water flow rate was approximately 6 gpm.

Since the temperature of the Siloxane was rapidly approaching its maximum working temperature of 600°F., the test was shut down for a re-check of equipment operation before the test materials became damaged.

Examination of the test parts showed that the O-ring seal at the propeller shaft was leaking and that Siloxane was entering the body of the electric motor. The motor was cleaned and checked for operation, a new O-ring was installed, and the test unit was again rigged for test.

Again, the cooling water flow was started and the propeller was energized and checked for correct operation. There was still some leakage of oil past the O-ring seal but it was decided to proceed. The flame was applied and temperature rises observed. A rapid rise in temperature of the Siloxane was again observed: before this temperature reached a critical point, the electric motor driving the propeller failed, and almost immediately thereafter the flame burned through the anode tip. At the time of burn-out, the temperature of the Siloxane, as observed by the probe, was below its safe high temperature limit.

The burned tip was removed from the test anode and was replaced with a tip having a wall thickness of 0.090 inch. Before re-testing an analysis was made of the data obtained so as to establish a logical next step.

Two things were apparent from the initial tests:

1) That the Force Thrust Convector, i.e., the coiled tube within the anode body, could not remove sufficient heat from the oil to reach thermal balance at temperature levels acceptable with the materials used.

2) That natural circulation of the oil in the cavity was inadequate to cool the anode tip, regardless of the capability of the coil to perform the heat transfer function.

However, sufficient data was not available to establish the amount of heat removed by the coil, or whether an increased rate of circulation of the Siloxane oil would improve the performance of the system.

Since a large capacity, external pump provided the most convenient method to control Siloxane oil flow within the body of the anode, a decision was made to use a reservoir of this oil, to pump it through the body, and to measure these effects on the system performance.

Accordingly, the propeller and motor were removed from the test anode and fluid lines were attached to provide a circulating loop from the reservoir to pump to anode and return to reservoir. Other conditions remained as in the initial test.

When the system described above was tested, it was noted that a continual temperature rise in the Siloxane oil occurred as before, although at a slower rate due to the larger volume of oil to be heated. The test was stopped and a cooling coil was inserted into the Siloxane circulating loop: this coil was immersed in a bucket containing a mixture of ice and water. The test was restarted, and the volume of water flow through the coil within the anode was reduced so as to achieve a measureable temperature rise and an evaluation of the heat being carried off by the coil. The test was again started, and data obtained as shown below. The anode temperature was read at the surface of the anode tip as close to the flame as the probe could be placed. This was approximately 1/2 inch from the flame.

Time Of Day	Anode Surface Temp. °F	Water Inlet To Anode Coil Temp. °F	Siloxane Oil Inlet Temp. °F	Water Outlet From Anode Coil Temp. °F	Siloxane Oil Outlet Temp. °F	Water Flow Rate GPM	Siloxane Flow Rate GPM	Acetylene Pressure PSI	Oxygen Pressure PSI
9:00	355	80	60	89	300	2.5	.22	7.0	7.7
9:30	355	78	60	87	300	2.5	.22	7.0	7.7
10:00	355	78	62	87	300	2.5	.22	7.0	7.7
10:30	358	78	61	87	300	2.5	.22	7.0	7.7
11:00	358	80	60	89	300	2.5	.22	7.0	7.7
11:30	360	80	60	89	300	2.5	.22	7.0	7.7
	Cycle Shut-down								
1:00	359	80	60	89	300	2.5	.22	7.0	7.7
1:30	360	80	60	89	300	2.5	.22	7.0	7.7
2:00	360	80	60	89	300	2.5	.22	7.0	7.7
2:30	360	80	60	89	300	2.5	.22	7.0	7.7
2:40	400	- Anode failure. Tip burned through. Hole approximately .001" diameter.							

Sample calculation for BTU/hr. absorbed by Siloxane and water:

$$\frac{\text{BTU}}{\text{Hr.}} = \left[CP_1 \times \frac{\text{Gal.}}{\text{Min.}} \times 1.34 \times 10^{-1} \frac{\text{Ft.}^3}{\text{Hr.}} \times \text{specific wgt.}_1 \times 60 \frac{\text{Min.}}{\text{Hr.}} \right] \times T_1 \ {}^{\circ}\text{F.}$$

$$+ \left[CP_2 \times \frac{\text{Gal.}}{\text{Min.}} \times 1.34 \times 10^{-1} \frac{\text{Ft.}^3}{\text{Gal.}} \times \text{specific wgt.}_2 \times 60 \frac{\text{Min.}}{\text{Hr.}} \right] \times T_2 \ {}^{\circ}\text{F.}$$

Where: CP_1 = specific heat of water = 1.0

CP_2 = specific heat of Siloxane = 3.4×10^{-1}

Specific weight of water = 62.4 lbs/ft.^{3}

Specific weight of Siloxane = 65.2 lbs/ft.^{3}

$$\frac{BTU}{Hr.} = \left[1.0 \times 2.5 \times 1.34 \times 62.4 \times 60\right] 9 + \left[.34 \times .22 \times 1.34 \times 62.4 \times 60\right] 240$$

$$\frac{BTU}{Hr.} = 11,288 + 9,006 = 20,294$$

Data Reduction:

The calculations show the following distribution of heat absorption by each of the two fluids, and the total, as follows:

Time Of Day	ΔT Water $^{\circ}F$	BTU Absorbed By Water	ΔT Siloxane $^{\circ}F$	Residual BTU Absorbed By Siloxane	Total BTU Absorbed
9:00	9	11,288	240	9,006	20,294
9:30	9	11,288	240	9,006	20,294
10:00	9	11,288	238	8,931	20,219
10:30	9	11,288	239	8,969	20,257
11:00	9	11,288	240	9,006	20,294
11:30	9	11,288	240	9,006	20,294
1:00	9	11,288	240	9,006	20,294
1:30	9	11,288	240	9,006	20,294
2:00	9	11,288	240	9.006	20,294
2:30	9	11,288	240	9.006	20,294

The data shows the total BTU unloaded by the system was below that necessary for a 20 KW lamp anode (27,300 BTU/Hr.) and failure occurred although excessive temperatures were not observed. It was concluded that the test condition of applying heat was too severe for the part: that with the point of the flame touching the anode tip, the localized heat could not be distributed by the metal and burn-through would occur.

It was also evaluated that there was insufficient surface area of the coiled tube: that additional area would improve the unloading of heat through this medium. A new coil was manufactured: it was sized as large as possible in diameter, and the tubing was partially flattened to attempt to get more coils in the anode body.

The insert thermocouple was forced to the lowest possible position in the anode in order to provide the best possible reading of anode body temperature near the tip.

A new tip with a wall thickness of 0.120 inch was placed on the test anode and the unit was again assembled for test. For this test the oxy-acetylene flame was lowered: the flame was about one inch long, with the tip of the blue flame approximately 1/4 inch below the anode tip. Data taken during this test was recorded as follows:

Time Of Day	Anode Surface Temp. °F.	Water Inlet Temp. °F.	Siloxane Inlet Temp. °F.	Water Outlet Temp. °F.	Siloxane Outlet Temp. °F.	Water Flow Rate GPM	Siloxane Flow Rate GPM	Acetylene Pressure PSI	Oxygen Pressure PSI
10:00	520	61	60	72	280	2.5	.22	7.0	7.7
10:30	520	60	60	71	280	2.5	.22	7.0	7.7
11:00	521	61	61	72	282	2.5	.22	7.0	7.7
11:30	520	61	60	72	280	2.5	.22	7.0	7.7
12:00	521	63	60	74	280	2.5	.22	7.0	7.7
12:30	522	63	60	74	280	2.5	.22	7.0	7.7
1:00	523	64	60	76	280	2.5	.22	7.0	7.7
1:30	521	64	60	76	280	2.5	.22	7.0	7.7
2:00	522	64	60	76	280	2.5	.22	7.0	7.7
2:30	520	64	60	76	280	2.5	.22	7.0	7.7
3:00	521	64	60	76	280	2.5	.22	7.0	7.7
3:30	522	64	60	76	280	2.5	.22	7.0	7.7
4:00	521	64	60	76	280	2.5	.22	7.0	7.7

Data Reduction:

The same calculations were used to ascertain distribution and extent of heat absorption by each of the two fluids:

Time Of Day	ΔT Water °F.	BTU Absorbed By Water	ΔT Siloxane °F.	Residual BTU Absorbed By Siloxane	Total BTU Absorbed
10:00	11	13,797	220	8,256	22,053
10:30	11	13,797	220	8,256	22,053
11:00	11	13,797	221	8,293	22,090
11:30	11	13,797	220	8,256	22,053
12:00	11	13,797	220	8,256	22,053
12:30	11	13,797	220	8,256	22,053
1:00	12	15,051	220	8,256	23,307
1:30	12	15,051	220	8,256	23,307
2:00	12	15,051	220	8,256	23,307
2:30	12	15,051	220	8,256	23,307
3:00	12	15,051	220	8,256	23,307
3:30	12	15,051	220	8,256	23,307
4:00	12	15,051	220	8,256	23,307

Examination of the anode tip at the end of the test run showed the tip to be in excellent condition. There was no sign of degradation or melting of the metal. There was present discoloration due to carbon residue deposited by the Acetylene.

It was also noted that during earlier tests, the water-cooled coil had unloaded $\frac{11,288}{20,294}$, or approximately 55.6% of the heat in the Siloxane fluid; that it was necessary to remove the remaining 44.4% by an external cooler. In the test with the revised anode cooling coil $\frac{13,797}{22,053}$, or approximately 62.5% of the heat was removed from the Siloxane.

In compliance with the general requirement that the test anode be started and stopped 25 times, the test was again started. Data is tabulated as follows:

Time Of Day	Anode Surface Temp. °F.	Water Inlet Temp. °F.	Siloxane Inlet Temp. °F.	Water Outlet Temp. °F.	Siloxane Outlet Temp. °F.	Water Flow GPM	Siloxane Flow GPM	Acetylene Pressure PSI	Oxygen Pressure PSI
10:00	580	62	60	74	288	2.5	.22	8.0	8.8
10:30	576	61	60	72	286	2.5	.22	8.0	8.8
11:00	578	61	60	72	286	2.5	.22	8.0	8.8
11:30	577	61	60	72	286	2.5	.22	8.0	8.8
12:00	580	61	62	72	287	2.5	.22	8.0	8.8
12:30	585	62	62	74	287	2.5	.22	8.0	8.8

1:00 Anode failure - Burn-through

Reduction of the above data tabulates as follows:

Time Of Day	ΔT Water °F.	BTU Absorbed By Water	ΔT Siloxane °F.	Residual BTU Absorbed By Siloxane	Total BTU Absorbed
10:00	12	15,051	288	8,556	23,607
10:30	11	13,797	226	8,481	22,278
11:00	11	13,797	226	8,481	22,278
11:30	11	13,797	226	8,481	22,278
12:00	11	13,797	225	8,444	22,241
12:30	12	15,051	225	8,444	23,495

In this test about $\frac{13,797}{22,278}$, or approximately 62% of the heat in the Siloxane was absorbed by the cooling coil.

At the time of failure, it was believed that sufficient data had been obtained to fulfill the purposes of the test, and that further testing would re-confirm information already on hand.

B.3 Discussion

Observation of the rapid failure of the first test anode when the forced circulation of the Siloxane ceased shows that natural circulation caused by heating of the fluid is far too low to permit unloading of high density heat by this mechanism. It is to be expected that the rate of circulation of the fluid is one of the critical items in the design: the circulation rate will affect the flow past the cooling coil in the anode thus its heat transfer efficiency, and it will also affect the rate of flow of coolant across the hot spot at the tip thus its capability to cool the critical area.

It was expected at the time the system was conceived that the high concentration of heat at the anode tip would induce a high rate of fluid circulation: the use of the propeller was to provide a safety factor which, hopefully, could be dispensed with. It was also expected that the rapid flow of fluid would tend to spread the heat load over a wider area: that operation of the metal at higher temperature would be of value in this spreading-out process.

The failure to achieve natural flow represents the most significant failure in the test program. However: the values of "Anode Body"

temperature measured in the early test are of doubtful value. At the time the test anode was last rebuilt, the thermocouple was placed lower, nearer the area of heat application. The marked rise represents measurement at a hotter location: it may or may not be at the point where the fluid temperature is hottest, but it probably is quite close to it. The significant item is that temperature of the fluid near the hot point is about at the maximum safe operating temperature for Siloxane: 600°F. This indicates that the flow rate of the Siloxane is about optimum from the point of view of maintenance of high temperature for heat transfer without exceeding allowable temperatures for the materials involved.

Use of air circulating through the coil could not change the results for the better: water temperature rise was nominal, and there was excess cooling capacity available if the heat transfer mechanism could use it. Whether air would or would not have equalled the water cooling rate is of little significance: the important item is that the Siloxane could not remove heat from the anode tip at a rate fast enough to prevent failure.

B.4 Conclusions

1) Spreading of heat from a "point" to an area through the techniques of operating at relatively high temperature and using natural conductance of the metal for spreading was unsuccessful. A better method appears to be to concentrate the cooling directly at the point of heat

application. This would seem to justify use of thin metal anode tips for most rapid conduction of heat with this design.

2) Methyl Phenol Siloxane can be used as a coolant provided the flow rate is sufficiently high.

3) The Dynamic Force Thrust Convection concept is not capable of adequately unloading heat to the extent, and within the geometric sizes, applicable to this program.

III. Recommendations for Future Action.

During the test program, and in an effort to get a satisfactory system test under way, a number of trials of various arrangements were experimented with. One of these was the use of dry ice as a coolant for the Siloxane: because the dry ice lowered the Siloxane inlet temperature to a level below that obtainable with a normal heat exchanger, dry ice was not used during the test run. The Siloxane was cooled only to room temperature.

However, an extended run of the test anode was made, using a higher concentrated heat application (estimated 40,000 BTU) than was used at any other time, during the time the Siloxane was cooled by the dry ice. Exact temperatures were not measured, but it is estimated that the inlet temperature was in the neighborhood of -30 to -40 degrees F.

After a run of several hours under these conditions there was no deterioration of the anode, while a less stringent test using normal inlet temperatures burned the anodes in a relatively short time.

It is the belief of this contractor that refrigeration of the coolant at the inlet may point the way to development of anodes capable of operating at higher lamp power than is presently possible. This does, of course, add complexity to the heat exchange system; if it also permits a rapid increase in lamp power rating without extensive development of higher temperature materials and improved coolants it may be a worthwhile area for further investigation.

Under the terms of contract DAAK02-68-C-0470 Aerospace Controls is required to deliver three (3) anodes with different tip configurations. As stated in the report three anode configurations were tested, but each failed. Therefore, instead of delivering destroyed material, an alternate delivery of three anodes is being made:

Each anode is designed the same as all others, and in accordance with attached drawing no. 2D1589. ACC recommends that these parts be tested starting with a laboratory mode similar to the tests described in this report and, if successful, continuing with a test in an experimental lamp.

The design of the anodes is such that the coolant is directed, with minimum contact with heated surfaces (to retain coolant low temperature), directly to the hot spot at the anode tip. The coolant is then directed through a center riser to an outlet: it is recommended that this outlet be connected to the anode outer chamber so that the same coolant provides a cool anode body. From the outer chamber the coolant is circulated through its outlet back to the reservoir.

The design embodies concepts which are based on beliefs that are widely held but which, at this time, have not been satisfactorily measured. Primary among these is the belief that temperatures experienced directly at the point of contact of the arc-center with the anode are substantially higher than has been suspected in the past; and that there are localized hot spots which tend to move

about over the anode surface and which cause intermittent melting of the metal at points. The intent of use of refrigerated coolant is to maintain <u>all</u> the metal of the anode tip at a temperature below freezing at all times: to thus prevent damage from melted, or burned, surfaces.

ACC would, of course, be interested in conducting such a test as a supplement to this program.

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List of Test Equipment

Thermocouples: Heat Technology Lab. Inc. P/N TC-J070602 Model CB-1.

Keithley Model 149 Millivoltmeter S/N 28491.

Thermo-Electric Co. Copper-Constantin Thermocouple wire, 18 gage.

Van Waters & Rogers Total Immersion Thermometer No. 61011-00-066.

range 0-300 °F in 2 °F increments (calibration of thermocouples).

One-quart container.

Timex watch with sweep-second hand.

Hypro high-temperature pump with Dayton 1.5 H.P. Motor.

Fischer & Porter Water Flow Meter, range 2-10 gpm, S/N 2235621.

Smith Welding Equipment Oxy-Acetylene Welder MW5 with Victor No. 5 cutting tip.

Methyl Phenol Siloxane, General Electric Versilube F-50 Silicone Fluid.

Pour Point = -100 °F

Operating range (air) = -100 °F to + 450 °F

Operating range (inert atmosphere) = -100 °F to over 600 °F

Specific heat = .034 BTU/lb./°F

Thermal conductivity = 0.087 BTU/hr.ft. °F

Specific gravity = 1.045

Viscosity = 2500 Centistokes

Expansion coefficient = 5.0×10^{-4} cc/cc/°F

Flash point = 550 °F min.

Fire point = 640 °F min.

Auto ignition = 850 °F

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T H I R D C L A S S M A I L

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HEAT TECHNOLOGY LAB. INC
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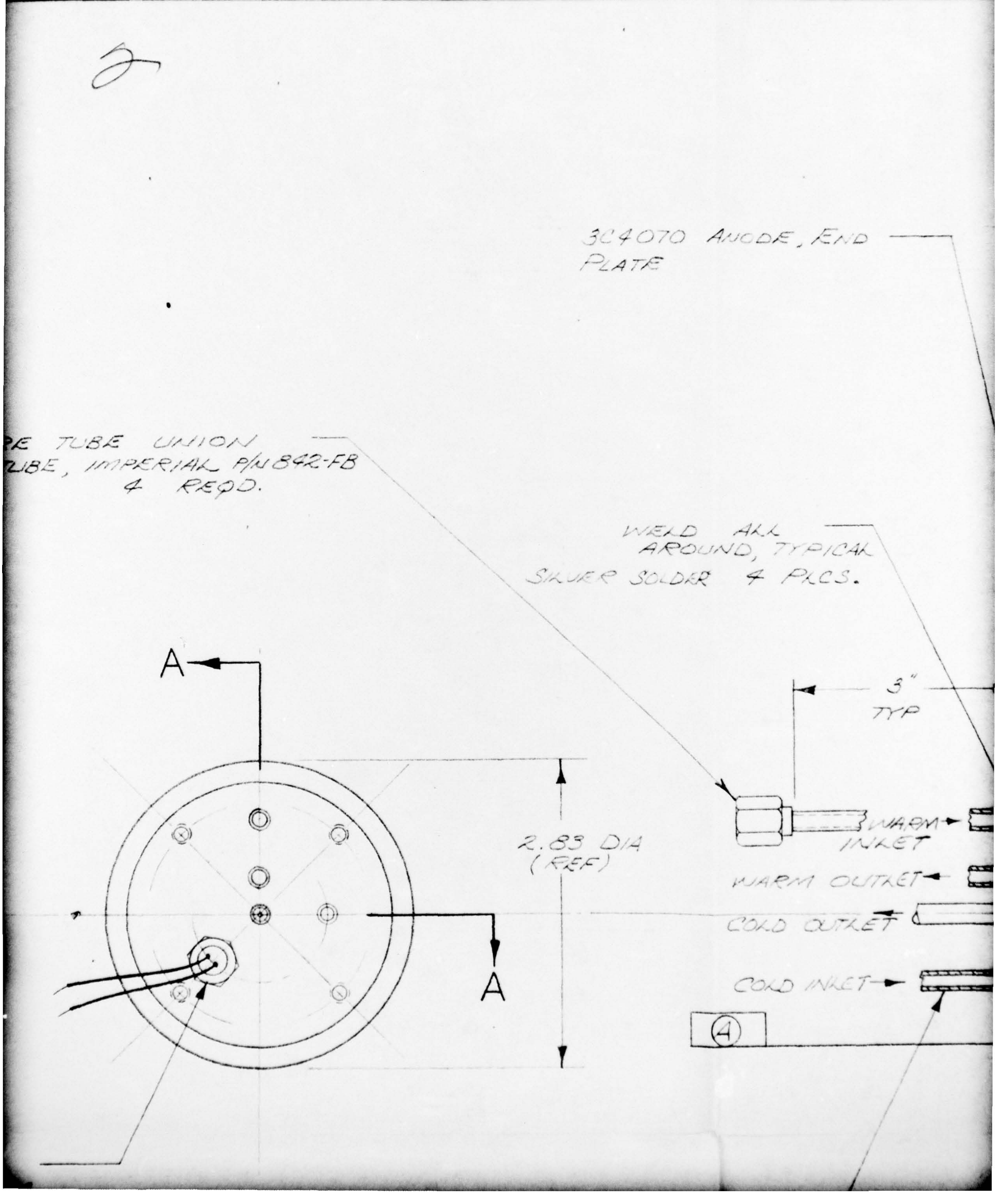

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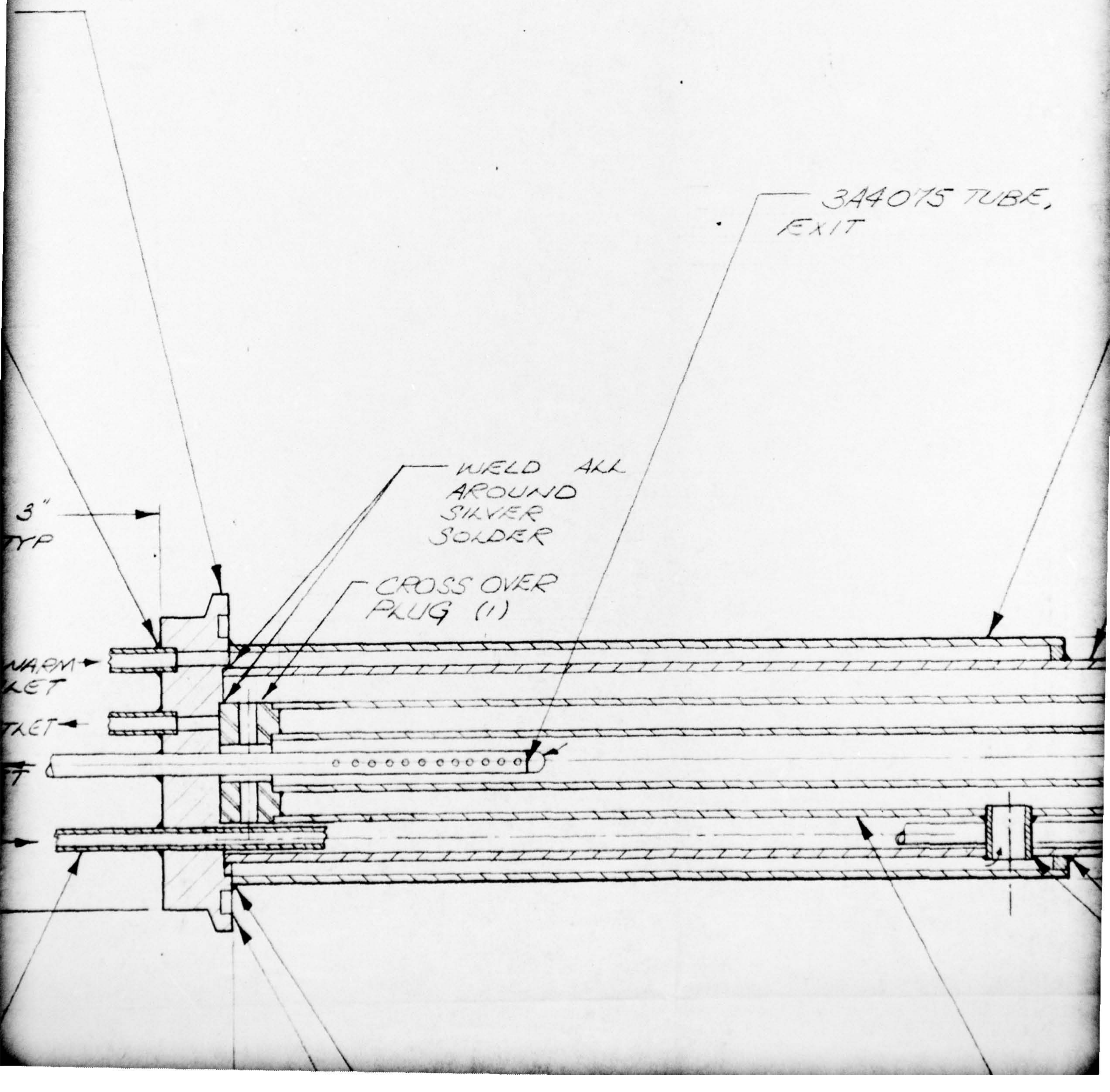
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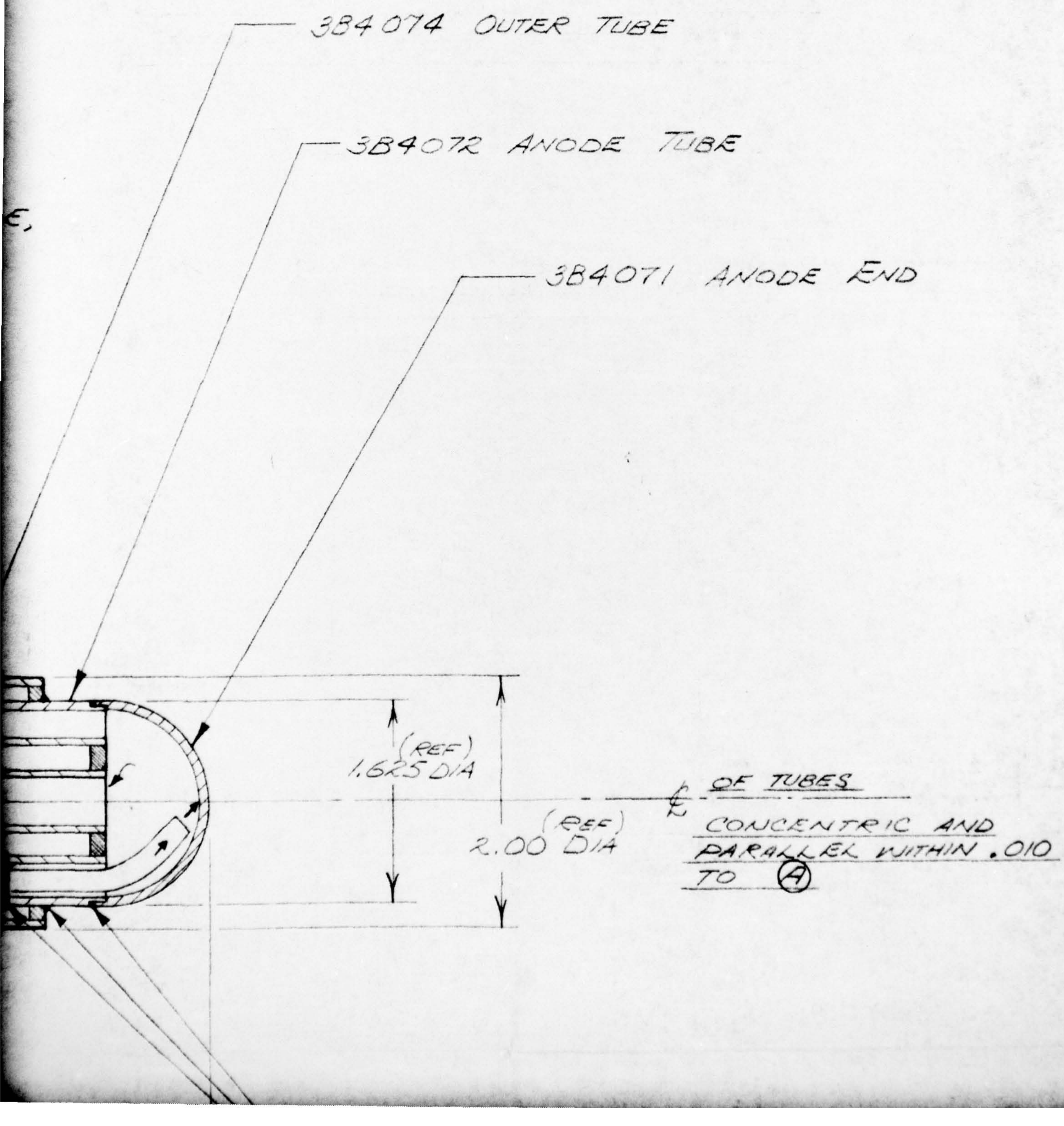
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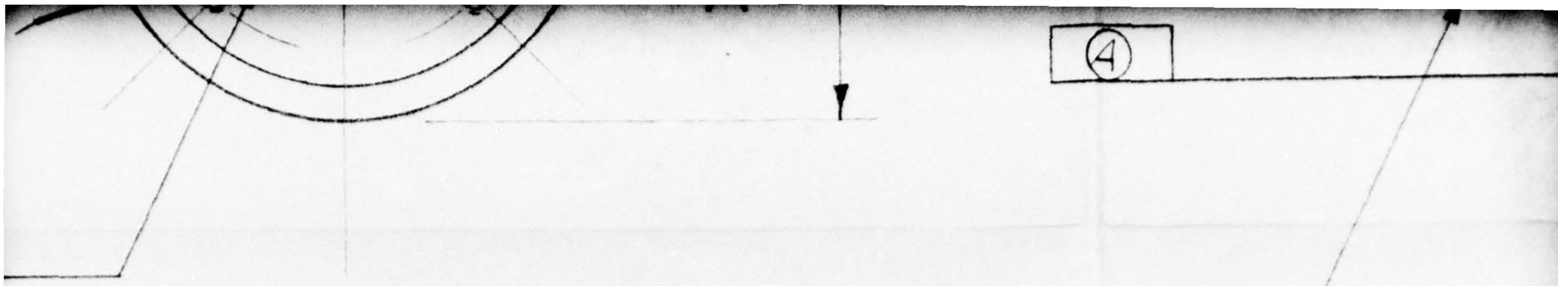

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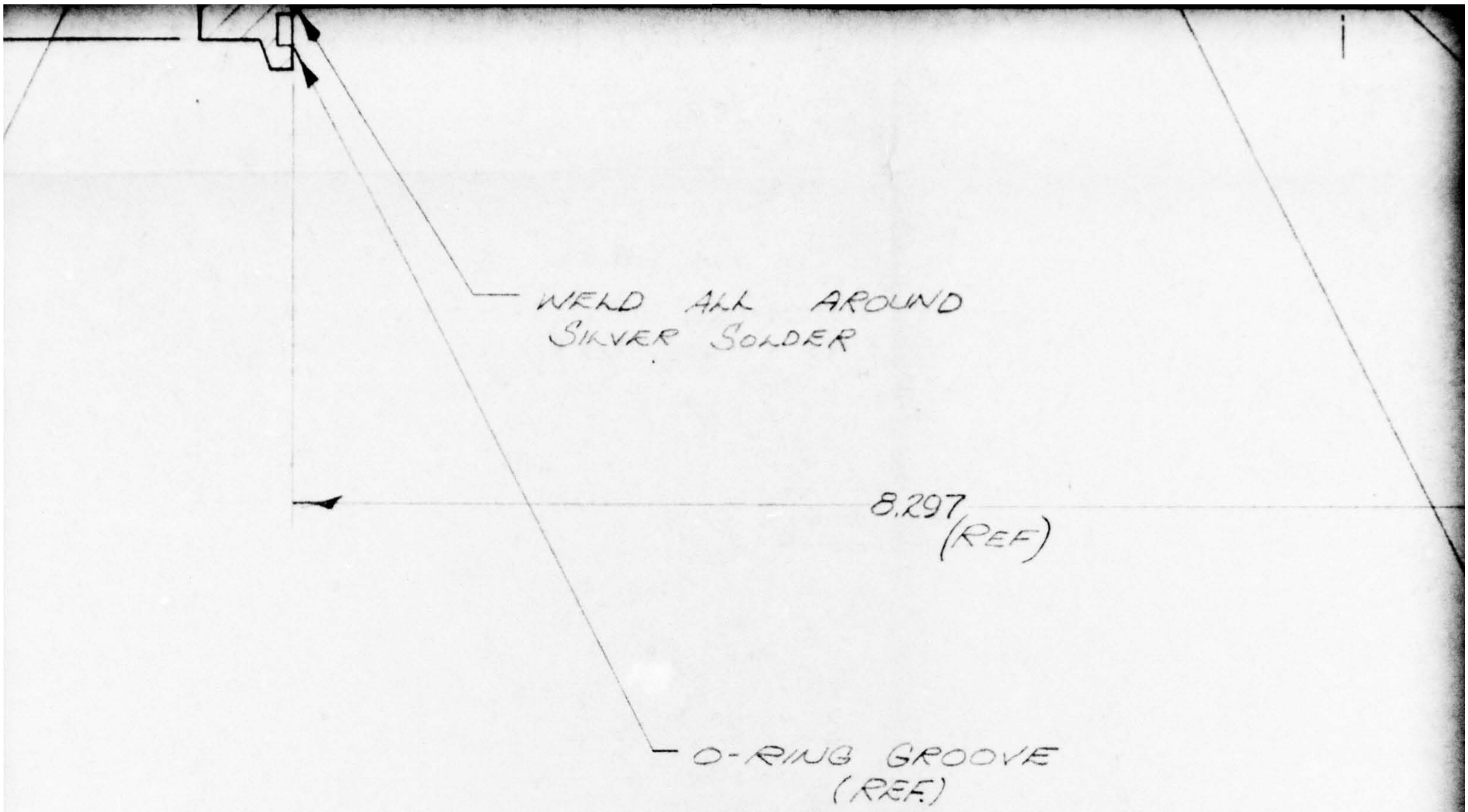


SECTION A-A

REPLACEM

NEXT ASSY	USED O
	P65

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES ON DRILLS ± .00
FRACTION ± 1/16
DECIMALS XX.030 XXX .010
ANGLES ± 1°
SURFACE 125

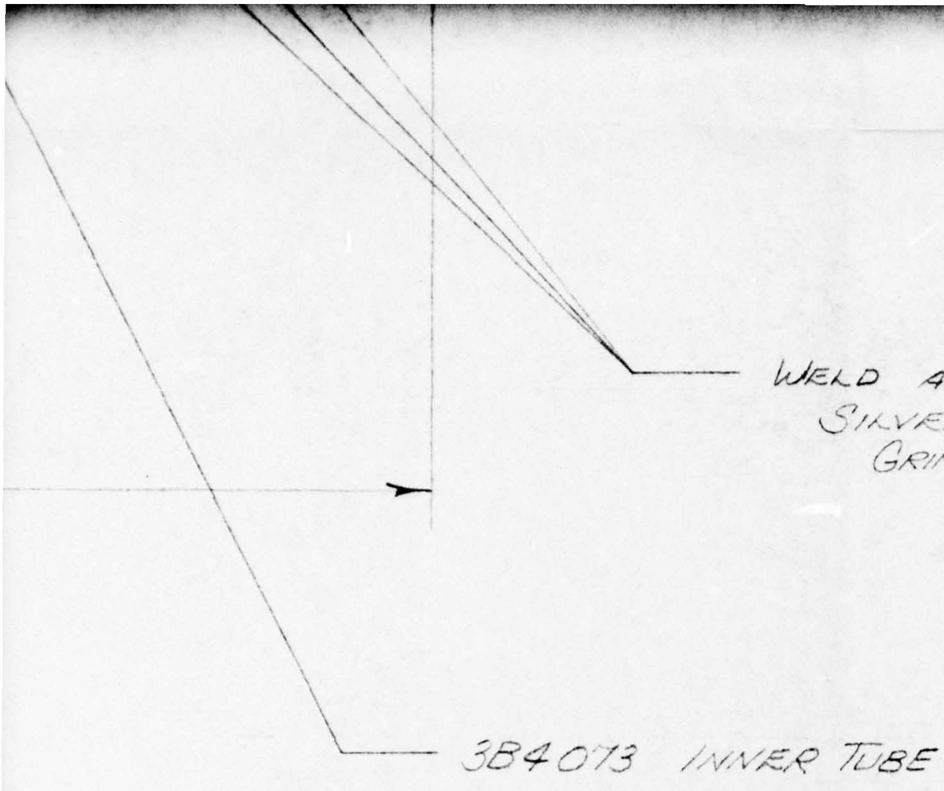
REMOVE BURRS AND
BREAK ALL SHARP EDGES

MATERIAL:

FINISH:

SYM	DESCRIPTION	DFTSM	DATE	APPROVAL

REVISIONS



REPLACEMENT FOR HANOVIA ANODE, (REF) 20KW

XT ASSY	USED ON	QTY REQD	SYM	DESCRIPTION	CODE IDENT	PART No.	SPECIFICATION	MATERIAL	UNIT WT
	P654	3 UNITS							

LIST OF MATERIALS

LESS OTHERWISE SPECIFIED:
MENSIONS ARE IN INCHES
LERANCES ON DRILLS ± .003
ACTION ± 1/16
CIMALS XX.030 XXX .010
GLES ± 1°
RFACE 125 √
MOVE BURRS AND
EAK ALL SHARP EDGES
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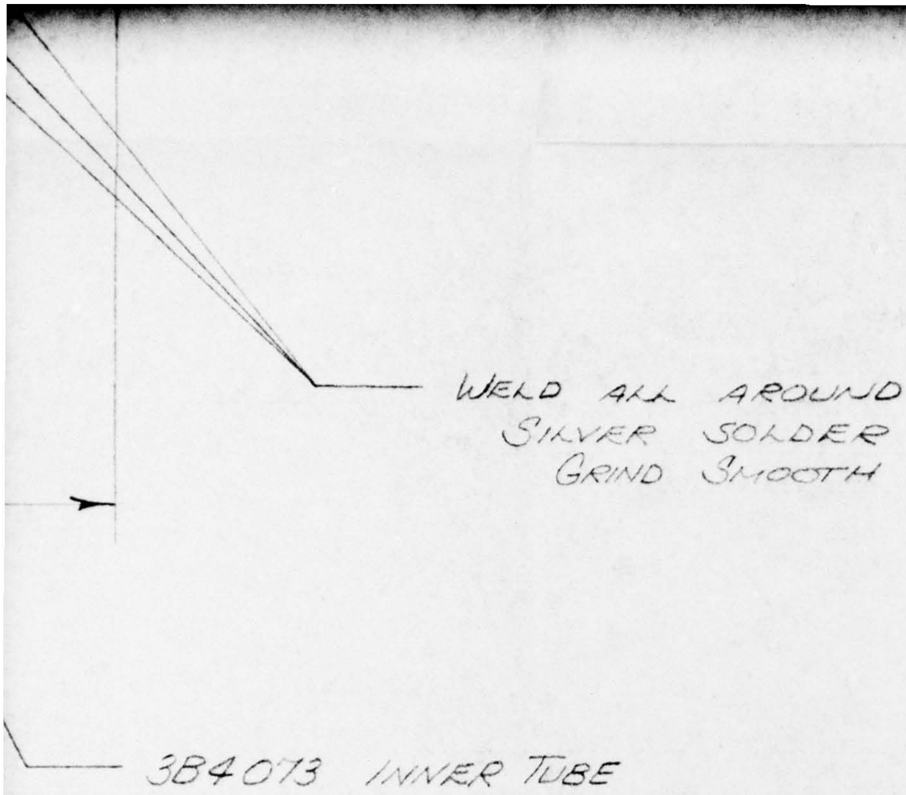
DRAFTSMAN: J. MIDDLETON DATE: 9/30/68
CHECKER:
ENGINEER: J.P. 10/3/68
ENGINEER:
PROD. APPROVAL:
APPROVAL: [signature] 10/3/68
APPROVAL:

AEROSPACE CONTROLS CORPORATIO
215 W. 131 ST. LOS ANGELES, CALIFO

TITLE: ASSEMBLY — ANODE ELECTROD (FABICATION)

CODE IDENT No.	SIZE	DRAWING No.
30292	D	2D158

SCALE FULL | UNIT WT. | SHEET 1



IT FOR HANOVIA ANODE, (REF) 20KW

QTY REQD	SYM	DESCRIPTION	CODE IDENT	PART No.	SPECIFICATION	MATERIAL	UNIT WT	ZONE	ITEM No.

3 UNITS

LIST OF MATERIALS

DRAFTSMAN: J. MIDDLETON — DATE: 9/30/68	AEROSPACE CONTROLS CORPORATION 215 W. 131 ST. LOS ANGELES, CALIFORNIA
CHECKER:	TITLE: ASSEMBLY— P654 ANODE ELECTRODE (FABICATION)
ENGINEER: J.R. 10/3/68	
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APPROVAL: [signature] 10/3/68	CODE IDENT No. 30292 — SIZE D — DRAWING No. 2D1589
APPROVAL:	SCALE FULL — UNIT WT. — SHEET 1 OF 1

9

Unclassified
Security Classification

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Final Technical Report,

5. AUTHOR(S) (First name, middle initial, last name)

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13. ABSTRACT

A Dynamic Force Thrust Convector Heat Exchanger, intended for use with air cooling of 20 KW short arc xernon lamps, was tested. The tests showed that thermal balance could not be achieved within tolerable limits for the materials used; that natural circulation of the thermal medium was below useful limits; and that the design does not distribute the high density heat loading characteristic of the lamp anode so as to afford an improved cooling method. It was found that refrigeration of coolant prior to introduction into the anode had a beneficial effect and is worth further investigation.

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DD FORM 1 NOV 65 1473 REPLACES DD FORM 1473, 1 JAN 64, WHICH IS OBSOLETE FOR ARMY USE.

Unclassified
Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
20 KW Xenon Lamp						
High density anode heat loading						
Heat transfer mechanism - anodes						
Analysis						
Nucleate boiling						
Hydrodynamic crisis						
Coolant characteristics						
Dynamic Force Thrust Convector						
Anode burning						
High temperature conductivity of anode						
Refrigerated coolant						